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Techno-economic survey of Organic Rankine Cycle (ORC) systems

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ABSTRACT

New heat conversion technologies need to be developed and improved to take advantage of the necessary increase in the supply of renewable energy. The Organic Rankine Cycle is well suited for these applications, mainly because of its ability to recover low-grade heat and the possibility to be implemented in decentralized lower-capacity power plants.

In this paper, an overview of the different ORC applications is presented. A market review is proposed including cost figures for several commercial ORC modules and manufacturers. An in-depth analysis of the technical challenges related to the technology, such as working fluid selection and expansion machine issues is then reported. Technological constraints and optimization methods are extensively described and discussed. Finally, the current trends in research and development for the next generation of Organic Rankine Cycles are presented.

Contents

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1. Introduction

The world energy consumption has risen to a level never reached before, releasing in the same process large quantities of CO_2 into the atmosphere. Current concerns over climate change call for measures to reduce greenhouse gases emissions, which will most likely include the following modifications of the current energy systems [1]:

(1) A decrease in the energy intensity of buildings and industry.
(2) A shift from fossil fuels toward electricity, e.g. for transportation and space heating.
(3) Clean power generation by a massive shift toward renewable energies, comprising wind energy, PV, CSP, biomass, geothermal and large hydro.
(4) A reinforcement of the grid capacity and inter-regional transmission lines to absorb daily and seasonal fluctuations.

Among the proposed solutions to fulfill these objectives, the Organic Rankine Cycle (ORC) technology can play a non-negligible role, in particular for objectives 1 and 3:

- It can have a beneficial effect on the energy intensity of industrial processes, mainly by recovering waste heat (i.e. heat that is otherwise lost). Installing an ORC to convert waste heat into electricity enables a better use of the primary energy. This approach is known as combined heat and power generation (CHP) through a bottoming cycle.
- It can have a positive effect on building consumptions, e.g. using CHP systems: since fossil fuels are able to generate high temperature levels, an ORC can take advantage of this high temperature to produce electricity. The low level temperature rejected by the ORC is still able to meet the needs of the building. This approach is known as combined heat and power generation through topping cycles.
- It can be used to convert renewable heat sources into electricity. This mainly includes geothermal, biomass and solar sources (CSP).
- During the transition toward electric vehicles, it can be used to increase the well-to-wheel efficiency by waste heat recovery on the exhaust gases, on the EGR and on the engine coolant.

Conceptually, the Organic Rankine Cycle is similar to a Steam Rankine Cycle in that it is based on the vaporization of a high pressure liquid which is in turn expanded to a lower pressure thus releasing mechanical work. The cycle is closed by condensing the low pressure vapor and pumping it back to the high pressure. Therefore, the Organic Rankine Cycle involves the same components as a conventional steam power plant (a boiler, a work-producing expansion device, a condenser and a pump). However, the working fluid is an organic compound characterized by a lower ebullition temperature than water and allowing power generation from low heat source temperatures.

In the rather new framework of decentralized conversion of low temperature heat into electricity, the ORC technology offers an interesting alternative, which is partly explained by its modular feature: a similar ORC system can be used, with little modifications, in conjunction with various heat sources. Moreover, unlike conventional power cycles, this technology allows for decentralized and small scale power generation.

These assets make the ORC technology more adapted than steam power to the conversion of renewable energy sources whose availability is generally more localized than that of fossil fuels, and whose temperature (e.g. in a solar collector or in a geothermal well) is lower than that of traditional fuels.

In this paper, an overview of the different ORC applications is presented. An in-depth analysis of the technical challenges related to this technology is proposed, such as working fluid or expansion machine issues. A market review is then given with cost figures for different commercial ORC modules and manufacturers, and the current trends in research and development are discussed.

2. ORC technology and applications

The layout of the Organic Rankine Cycle is somewhat simpler than that of the steam Rankine cycle: there is no water–steam drum connected to the boiler, and one single heat exchanger can be used to perform the three evaporation phases: preheating, vaporization and superheating. The variations of the cycle architecture are also more limited: reheating and turbine bleeding are generally not suitable for the ORC cycle, but a recuperator can be installed as liquid preheater between the pump outlet and the expander outlet, as illustrated in Fig. 1. This allows reducing the amount of heat needed to vaporize the fluid in the evaporator.

The simple architecture presented in Fig. 1 can be adapted and optimized depending on the target application. The main applications are briefly described in the following sections. Although this review only focuses on state-of-the art commercially available ORC plants, it should be noted that some prospective advanced applications for Organic Rankine Cycles are currently being studied, mainly in the form of prototypes or proof-of-concepts. These innovative applications include:

- Solar pond power systems, in which the ORC system takes advantage of temperature gradients in salt-gradient solar ponds [2].
- Solar ORC-RO desalination systems, where the ORC is used to drive the pump of a reverse-osmosis desalination plant [2,3].
- Ocean thermal energy conversion systems, utilizing the temperature gradients (of at least 20 °C) in oceans to drive a binary cycle [2].
- Cold production, where the shaft power of the ORC system is used to drive the compressor of a refrigeration system. Note that

Nomenclature

D	diameter (m)
h	specific enthalpy (J/(kg K))
$\dot{M}$	mass flow rate (kg/s)
N	rotating speed (Hz)
p	pressure (Pa)
pinch	pinch point value (K)
$\dot{Q}$	heat power (W)
r	ratio (–)
$r_{v,in}$	internal built-in volume ratio (–)
T	temperature (°C)
U	peripheral speed (m/s)
V	velocity (m/s)
v	specific volume (m^3/kg)
$\dot{W}$	electrical or mechanical power (W)

Greek symbols

ε	effectiveness
η	efficiency
φ	filling factor
ρ	density (kg/m^3)

Subscripts and superscripts

amb	ambient
c	critical
cd	condenser
el	electrical
em	eletromechanical
ev	evaporator
ex	exhaust
exp	expander
in	internal
mech	mechanical
ncg	non-condensing gases
pp	pump
sc	subcooling
su	supply
tp	two-phase
tot	total

Acronyms

CHP	combined heat and power
CSP	concentrated solar power
EGR	exhaust gas recirculation
GHG	green house gases
GWP	global warming potential
ICE	internal combustion engine
NPSH	net positive suction head
ODP	ozone depleting potential
ORC	Organic Rankine Cycle
WHR	waste heat recovery

this layout can also be used to produce heat with a COP > 1 if the ORC is coupled to a heat pump [3].

For an overview of these more innovative and prospective applications, the interested reader can refer to [2,3].

2.1. Biomass combined heat and power

Biomass is widely available in a number of agricultural or industrial processes such as wood industry or agricultural waste. Among other means, it can be converted into electricity by combustion to obtain heat, which is in turn converted into electricity through a thermodynamic cycle. The cost of biomass is significantly lower than that of fossil fuels. Yet, the investment necessary to achieve clean biomass combustion is more important than for classic boilers. For small decentralized units, the generation cost of electricity is not competitive and combined heat and power generation is required to ensure the profitability of the investment. Therefore, in order to achieve high energy conversion efficiency, biomass CHP plants are usually driven by the heat demand rather than by the electricity demand [4].

The possibility to use heat as a by-product is an important asset of biomass ORCs, highlighting the importance of a local heat demand, which can be fulfilled e.g. by industrial processes (such

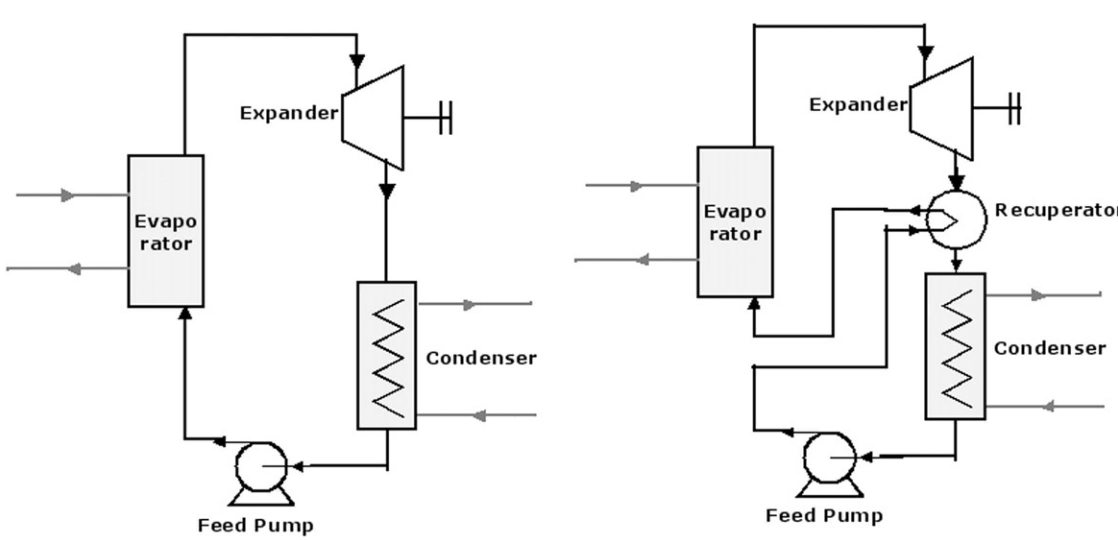


Fig. 1. Schematic view of an ORC with (right) and without (left) recuperator.

as wood drying) or space heating (usually district heating) [5]. Since heat is relatively difficult to transport across long distances, biomass CHP plants are most of the time limited to 6–10 MW thermal power, corresponding to 1–2 MW electrical power. This excludes traditional steam cycles that are not cost-effective in this power range (this particular point will be further developed in Section 4).

Simplified diagrams of such cogeneration systems are proposed in Figs. 2 and 3: heat from the combustion is transferred from the flue gases to the heat transfer fluid (thermal oil) in two heat exchangers, at a temperature varying between 150 and 320 °C. The heat transfer fluid is then directed to the ORC loop to evaporate the working fluid, at a temperature slightly lower than 300 °C. Next, the evaporated fluid is expanded, passes through a recuperator to preheat the liquid and is finally condensed at a temperature around 90 °C. The condenser is used for hot water generation.

For the particular example of Fig. 2, although the electrical efficiency of the CHP system is limited (18%), the overall efficiency of the system is 88%, which is much higher than that of centralized power plants, in which most of the residual heat is lost.

To reduce heat losses in the flue gases, these gases must be cooled down to the lowest possible temperature, insofar as the acid dew point is not reached. To achieve this, two heat transfer loops are used: a high temperature loop and a low temperature loop. The low temperature loop is installed after the high temperature loop on the flue gases to reduce their outlet temperature (Fig. 3).

The main competing technology for electricity generation from solid biofuels is biomass gasification: in this technology, biomass is transformed into a synthetic gas composed mainly of H_2, CO, CO_2 and CH_4. This synthetic gas is treated and filtered to eliminate solid particles, and is finally burned in an internal combustion engine or in a gas turbine.

When comparing the technology and the costs of biomass CHP using an ORC with gasification, it can be shown that gasification involves higher investment costs (about 75%) and higher operation and maintenance costs (about 200%). On the other hand, gasification yields a higher power-to-thermal ratio, which makes its exploitation more profitable [6]. It should also be noted that ORC is a well-proven technology, while gasification plants in actual operation are mostly prototypes for demonstration purposes.

2.2. Geothermal energy

Geothermal heat sources are available over a broad range of temperatures, from a few tens of degrees up to 300 °C. The actual technological lower bound for power generation is about 80 °C: below this temperature the conversion efficiency becomes too small and geothermal plants are not economical. Table 1 indicates the potential for geothermal energy in Europe and shows that this potential is very high for low temperature sources.

To recover heat at an acceptable temperature, boreholes must generally be drilled in the ground, for the production well and for the injection well (cfr. Fig. 4). The hot brine is pumped from the former and injected into the latter at a lower temperature. Depending on the geological formation, boreholes can be several thousand meters deep, requiring several months of continuous

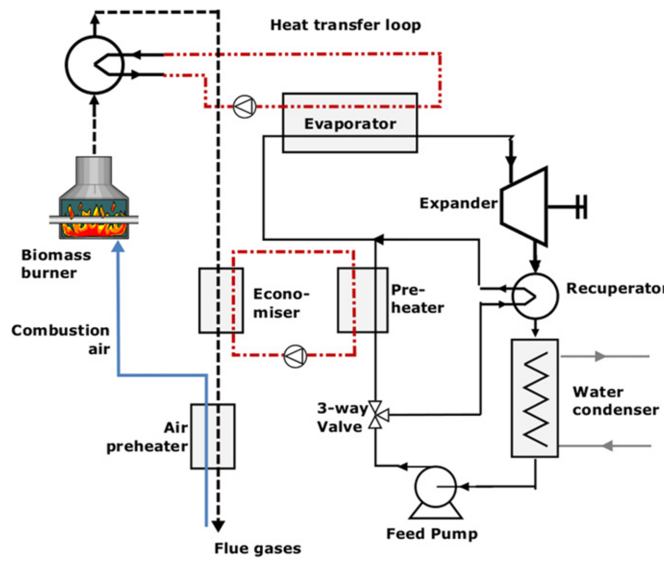


Fig. 3. Working principle of a biomass CHP ORC system.

Table 1
Potential for geothermal energy in Europe for different heat source temperature ranges [9].

Temperature (°C)	MWth	MWe
65–90	147,736	10,462
90–120	75,421	7503
120–150	22,819	1268
150–225	42,703	4745
225–350	66,897	11,150

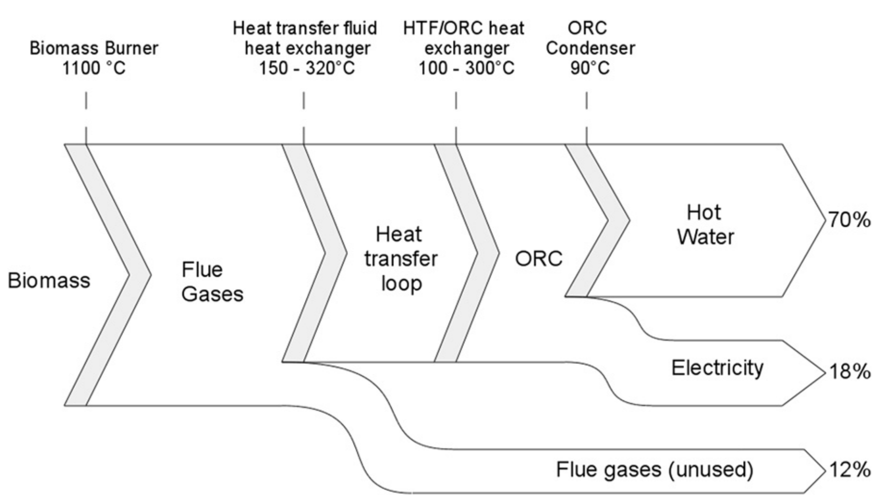


Fig. 2. Energy flow as a function of the conversion temperatures in a CHP ORC system.

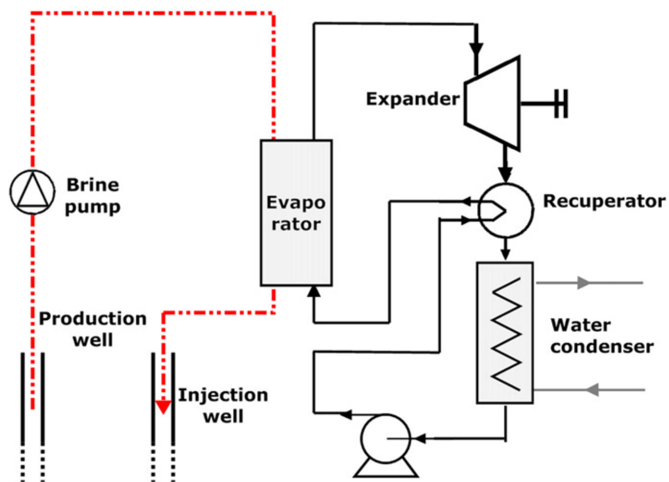


Fig. 4. Working principle of a geothermal ORC system.

work. According to Kranz [7], this leads to a high share of drilling cost in the total investment cost (up to 70%) of a geothermal ORC plant. Lazzaretto et al. [8] reports a much more moderate share of 15.6% for an Italian geothermal binary cycle.

Low-temperature geothermal ORC plants are also characterized by relatively high auxiliary consumption: the pumps consume from 30% up to more than 50% of the gross output power [9]. The main consumer is the brine pump that has to circulate the brine over large distances and with a significantly high flow rate. The working fluid pump consumption is also higher than in higher temperature cycles, because the ratio between pump consumption and turbine output power ("back work ratio") increases with decreasing evaporation temperature (see Section 8.2).

Higher temperature (> 150 °C) geothermal heat sources enable combined heat and power generation: the condensing temperature is set to a higher level (e.g. 60 °C), allowing the cooling water to be used for district heating. In this case, the overall energy recovery efficiency is increased, but at the expense of a lower electrical efficiency.

2.3. Solar power plants

Concentrating solar power is a well-proven technology: the sun is tracked and its radiation reflected onto a linear or punctual collector, transferring heat to a fluid at high temperature. This heat is then used in a power cycle to generate electricity. The three main concentrating solar power technologies are the parabolic dish, the solar tower, and the parabolic trough. Parabolic dishes and solar towers are punctual concentration technologies, leading to a higher concentration factor and to higher temperatures. The most appropriate power cycles for these technologies are the Stirling engine (for small-scale plants), the steam cycle, or even the combined cycle (for solar towers).

Parabolic troughs work at a lower temperature (300–400 °C) than point-focused CSP systems. Up to now, they were mainly coupled to traditional steam Rankine cycles for power generation [10]. They are subject to the same limitations as in geothermal or biomass power plants: steam cycles require high temperatures, high pressures, and therefore larger installed power to be profitable.

Organic Rankine Cycles are a promising technology to decrease investment costs at small scale: they can work at lower temperatures, and the total installed power can be scaled down to the kW levels. The working principle of such a system is presented in Fig. 5. Technologies such as Fresnel linear concentrators [11] are

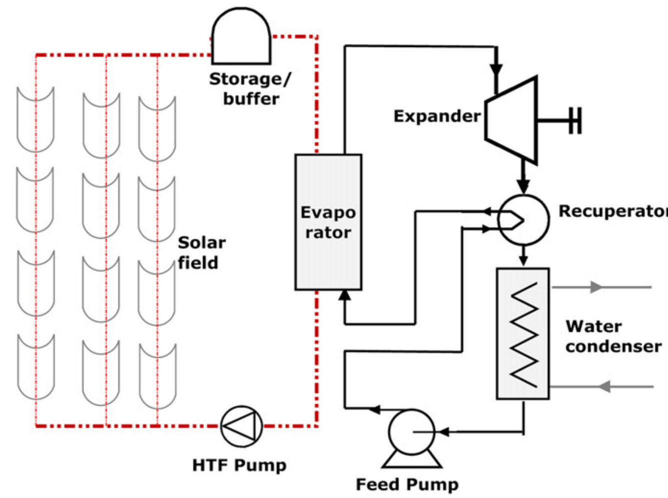


Fig. 5. Working principle of a solar ORC system.

particularly suitable for solar ORCs since they require a lower investment cost, but work at lower temperature.

Up to now, very few CSP plants using ORC are available on the market:

- A 1 MWe concentrating solar power ORC plant was completed in 2006 in Arizona. The ORC module uses n-pentane as the working fluid and shows an efficiency of 20%. The overall solar to electricity efficiency is 12.1% at the design point [12].
- A 100 kWe plant was commissioned in 2009 in Hawaii by Electratherm. The heat transfer fluid temperature in the collectors is about 120 °C.
- Some very small-scale systems are being studied for remote off-grid applications, such as the proof-of-concept kWe system developed for rural electrification in Lesotho by "STG International" [13].

2.4. Waste heat recovery

2.4.1. Heat recovery on mechanical equipment and industrial processes

Many applications in the manufacturing industry reject heat at relatively low temperature. In large-scale plants, this heat is usually overabundant and often cannot be reintegrated entirely on-site or used for district heating. It is therefore rejected to the atmosphere.

This causes two types of pollution: pollutants (CO_2, NO_x, SO_x, HC) present in the flue gases generate health and environmental issues; heat rejection perturbs aquatic equilibriums and has a negative effect on biodiversity [14].

Recovering waste heat mitigates these two types of pollution. It can moreover generate electricity to be consumed on-site or fed back to the grid. In such a system, the waste heat is usually recovered by an intermediate heat transfer loop and used to evaporate the working fluid of the ORC cycle. A potential of 750 MWe is estimated for power generation from industrial waste heat in the US, 500 MWe in Germany and 3000 MWe in Europe (EU-12) [15].

Some industries present a particularly high potential for waste heat recovery. One example is the cement industry, where 40% of the available heat is expelled through flue gases. These flue gases are located after the limestone preheater or in the clinker cooler, with temperatures varying between 215 °C and 315 °C [16].

CO_2 emissions from the cement industry amount for 5% of the total world GHG emissions, and half of it is due to the combustion of fossil fuels in the kilns [14]. Other examples include the iron and steel industries (10% of the GHG emission in China for example), refineries and chemical industries.

2.4.2. Heat recovery on internal combustion engines

An Internal Combustion Engine (ICE) only converts about one-third of the fuel energy into mechanical power on typical driving cycles: a typical 1.4 l Spark Ignition ICE, with a thermal efficiency ranging from 15% to 32%, releases 1.7–45 kW of heat through the radiator (at a temperature close to 80–100 °C) and 4.6–120 kW via the exhaust gas (400–900 °C) [17].

The heat recovery Rankine cycle system (both organic and steam based) is an efficient means for recovering heat (in comparison with other technologies such as thermo-electricity and absorption cycle air-conditioning). The concept of applying a Rankine cycle to an ICE is not new and the first technical developments appeared after the 1970 energy crisis. For instance, Mack Trucks [18] designed and built a prototype of such a system operating on the exhaust gas of a 288 HP truck engine. A 450 km on-road test demonstrated the technical feasibility of the system and its economical interest: a reduction of 12.5% in the fuel consumption was reported. Systems developed today differ from those of 1970 because of advances in the development of expansion devices and the broader choice of working fluids. However, currently, no commercial Rankine cycle solution is available.

Most of the systems under development recover heat from the exhaust gases and from the cooling circuit [19]. By contrast, the system developed by [20] only recovers heat from the cooling circuit. An additional potential heat source is the exhaust gas recirculation (EGR) and charge air coolers, in which non-negligible amounts of waste heat are dissipated.

The expander output can be mechanical or electrical. With a mechanical system, the expander shaft is directly connected to the engine drive belt, with a clutch to avoid power losses when the ORC power output is too low. The main drawback of this configuration is the imposed expander speed: this speed is a fixed ratio of the engine speed and is not necessarily the optimal speed for maximizing cycle efficiency. In the case of electricity generation, the expander is coupled to an alternator, used to refill the batteries or supply auxiliary utilities, such as the air conditioning. It should be noted that current vehicle alternators show a quite low efficiency (about 50–60%), which reduces the ORC output power.

As for the expander, the pump can be directly connected to the drive belt, to the expander shaft, or to an electrical motor. In the latter case, the working fluid flow rate can be controlled independently, which makes the regulation of such a system much easier.

The control of the system is particularly complex due to the (often) transient regime of the heat source. However, optimizing the control is crucial to improve the performance of the system. It is generally necessary to control both the pump speed and the expander speed to maintain the required conditions (temperature, pressure) at the expander inlet [21].

Another technical constraint is the heat rejection capacity. The size of the front heat exchanger (either an air-cooled condenser or the radiator connected to a water-cooled condenser) is limited by the available space and depends on the presence of an engine radiator, and possibly a charge air cooler, an EGR cooler or an air-conditioning condenser. The system should be controlled such that the rejected heat remains within the cooling margin, defined as the cooling capacity without operating the cooling fans. Otherwise, fan consumption can sharply reduce the net power output of the system [22].

The performance of recently developed prototypes of Rankine cycles is promising: the system designed by Honda [23] showed a maximum cycle thermal efficiency of 13%. At 100 km/h, this yields a cycle output of 2.5 kW (for an engine output of 19.2 kW) and represents an increase of the engine thermal efficiency from 28.9% to 32.7%.

A competing technology under research and development is the thermoelectric generator (TEG), which is based on the Seebeck effect: its main advantages are a substantially lower weight than the ORC system, and the absence of moving parts. Major drawbacks are the cost of materials (which include rare earth metals) and the low achieved efficiency.

3. ORC manufacturer and market evolution

ORC manufacturers have been present on the market since the beginning of the 1980s. They provide ORC solutions in a broad range of power and temperature levels, as shown in Table 2. Note that only manufacturers with several commercial references have been detained in this survey.

The three main manufacturers in terms of installed units and installed power are Turboden (Pratt & Whitney) (45% of installed units worldwide, 8.6% of cumulated power), ORMAT (24% of installed units, 86% of cumulated power) and Maxxtec (23% of installed units, 3.4% of cumulated power) [24]. The large share

Table 2
Non-exhaustive list of the main ORC manufacturers.
Sources: Manufacturers websites; [24–32].

Manufacturer	Applications	Power range [kWe]	Heat source temperature [°C]	Technology
ORMAT, US	Geo., WHR, solar	200–70,000	150–300	Fluid : *n*-pentane and others, two-stage axial turbine, synchronous generator
Turboden, Italy	Biomass-CHP, WHR, Geo.	200–2000	100–300	Fluids : OMTS, Solkatherm, Two-stage axial turbines
Adoratec/Maxxtec, Germany	Biomass-CHP	315–1600	300	Fluid: OMTS
Opcon, Sweden	WHR	350–800	< 120	Fluid: Ammonia, Lysholm Turbine
GMK, Germany	WHR, Geo., Biomass-CHP	50–5000	120–350	3000 rpm Multi-stage axial turbines (KKK)
Bosch KWK, Germany	WHR	65–325	120–150	Fluid: R245fa
Turboden PureCycle, US	WHR, Geo.	280	91–149	Radial inflow turbine, Fluid: R245fa
GE CleanCycle	WHR	125	> 121	Single-state radial inflow turbine, 30,000 rpm, Fluid: R245fa
Cryostar, France	WHR, Geo.	n/a	100–400	Radial inflow turbine, Fluids: R245fa, R134a
Tri-o-gen, Netherlands	WHR	160	> 350	Radial turbo-expander, Fluid: Toluene
Electratherm, US	WHR, Solar	50	> 93	Twin screw expander, Fluid: R245fa

of ORMAT in the cumulated power is explained by its focus on large-scale, low temperature geothermal binary plants.

It should be noted that, in addition to the manufacturers listed in Table 2, many companies are entering the ORC market with low-capacity units, e.g. for micro-CHP or for WHR on IC engine exhaust gases. However, these companies have not yet reached sufficient technical maturity for large-scale competitive commercialization [25].

The ORC market is growing rapidly. Since the first installed commercial ORC plants in the 1970s, an almost-exponential growth has been stated, as visualized in Fig. 6, where the evolution of installed power and the number of plants in operation, based on a compilation of manufacturer data, is depicted.

Fig. 6 also reveals that the ORC is a mature technology for waste heat recovery, biomass CHP and geothermal power, but is still very uncommon for solar applications. Moreover, systems are mainly installed in the MW power range and very few ORC plants exist in the kW power range.

The variety of ORC modules is large and can be categorized according to unit size, type of technology, and target application. In Fig. 7 some typical ORC module costs, for different applications, are plotted as a function of their size. Note that the provided costs are indicative only and partially collected from a non-exhaustive set of ORC manufacturers and from scientific publications [26,33–35]. The scattering in the data is due to different prices for different manufacturers, different market strategies, different integration costs, etc. Therefore, individual costs should not be generalized, but are given merely to illustrate the general trend of system prices relative to the output power. Fig. 7 indicates that, for a given target application, the cost tends to decrease when the output power increases. Lowest costs are reported for waste heat recovery applications, while geothermal and CHP plants exhibit higher total cost. Total cost differs from module cost in that it includes engineering, buildings, boiler (in case of CHP), process integration, etc., and can amount to two to three times the module cost. These surplus costs should therefore never be neglected when evaluating the economics of an ORC plant.

4. Comparison with the steam Rankine cycle

This section provides a summary of the advantages and drawbacks of the ORC technology. The interested reader can refer to previous publications by some of the authors for a more detailed analysis [36,37].

Fig. 8 shows in the *T–s* diagram the saturation curves of water and of a few typical organic fluids used in ORC applications. Two main differences can be stated: (1) the slope of the saturated vapor curve (right curve of the dome) is negative for water, while the curve is much closer to vertical for organic fluids. As a consequence, the limitation of the vapor quality at the end of the expansion process disappears in an ORC cycle, and there is no need to superheat the vapor before the turbine inlet. (2) The entropy difference between saturated liquid and saturated vapor is much smaller for organic fluids. Hence, the vaporization enthalpy is smaller. Therefore, to take up equal thermal power in the evaporator, the organic working fluid mass flow rate must be much higher than for water, leading to higher pump consumption.

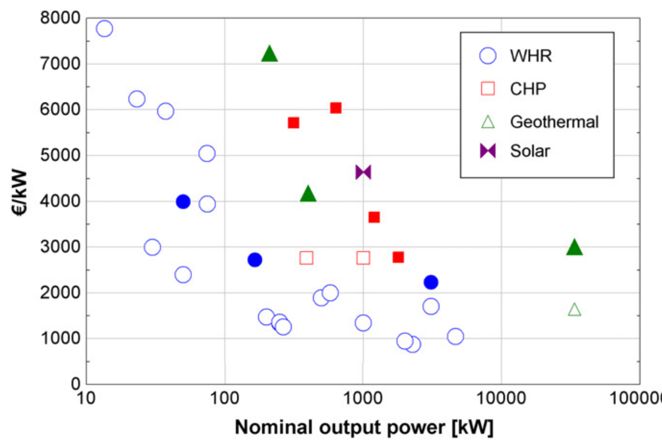


Fig. 7. Module (empty dots) and total (plain dots) cost of ORC systems depending on the target application and on the net electrical power.

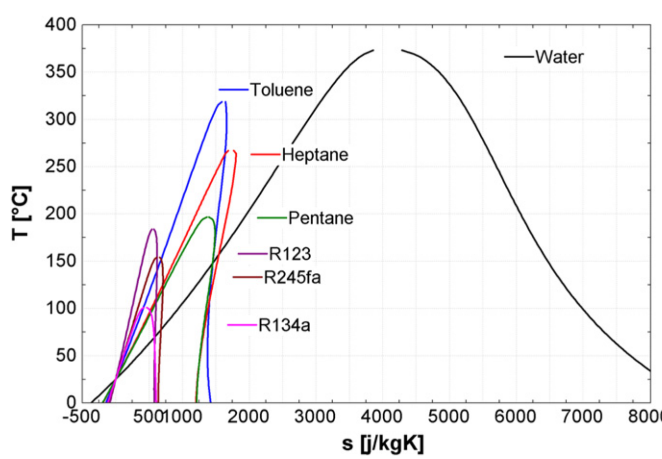


Fig. 8. *T–s* diagram of water and various typical ORC fluids.

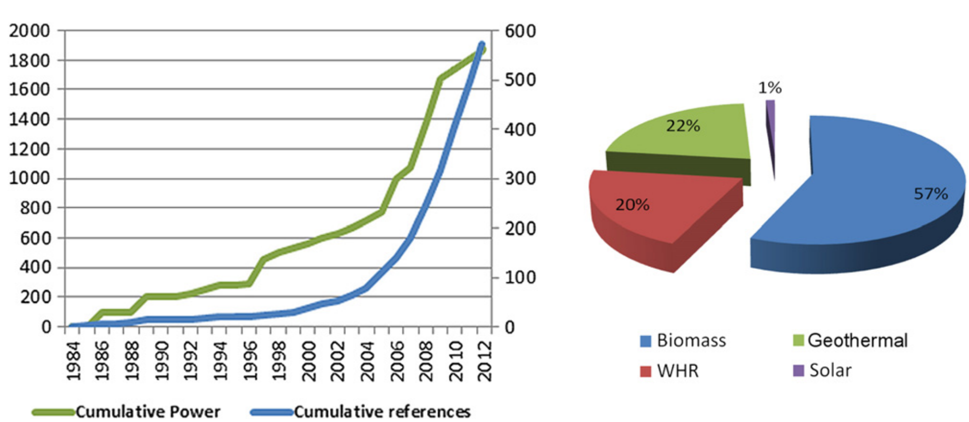


Fig. 6. Market evolution (left) and share of each application in terms of number of units (right) Data Source: [24].

The main differences between ORCs and steam cycles are the following:

- *Superheating*. As previously stated, organic fluids usually remain superheated at the end of the expansion. Therefore, there is no need for superheating in ORC cycles, contrary to steam cycles. The absence of condensation also reduces the risk of corrosion on the turbine blades, and extends its lifetime to 30 years instead of 15–20 years for steam turbines [14].
- *Low temperature heat recovery*. Due to the lower boiling point of a properly selected organic working fluid, heat can be recovered at much lower temperature (e.g. with geothermal sources).
- *Components size*. In a steam cycle, the fluid density is extremely low in the low-pressure part of the cycle. Since pressure drops increase with the square of the fluid velocity, the high volume flow rate necessitates an increase in the hydraulic diameter of the piping and the size of the heat exchangers. Similarly, the turbine size is roughly proportional to the volume flow rate.
- *Boiler design*. ORC cycles enable the use of once-trough boilers, which avoids steam drums and recirculation. This is due to the relatively smaller density difference between vapor and liquid for high molecular weight organic fluids. In contrast, the low vapor density in steam boilers can generate very different heat transfer and pressure drop characteristics between liquid water and steam. Complete steam evaporation in a single tube must therefore be avoided.
- *Turbine inlet temperature*. In steam Rankine cycles, due to the superheating constraint, a temperature higher than 450 °C is required at the turbine inlet to avoid droplets formation during the expansion. This leads to higher thermal stresses in the boiler and on the turbine blades and to a higher cost.
- *Pump consumption*. Pump consumption is proportional to the liquid volume flow rate and to the pressure difference between outlet and inlet. It can be expressed in terms of the Back Work Ratio (BWR), which is defined as the pump consumption divided by the turbine output power. In a steam Rankine cycle, the water flow rate is relatively low and the BWR is typically 0.4%. For a high temperature ORC using toluene, the typical value is 2–3%. For a low temperature ORC using HFC-134a, values higher than 10% are typical. Generally speaking, the lower the critical temperature, the higher the BWR.
- *High pressure*. In a steam cycle, pressures of about 60–70 bar and thermal stresses increase the complexity and the cost of the steam boiler. In an ORC, pressure generally does not exceed 30 bar. Moreover, the working fluid is not evaporated directly by the heat source (e.g. a biomass burner) but via an intermediary heat transfer loop. This makes the heat recovery easier since thermal oil can be at ambient pressure, and the requirement of an on-site steam boiler operator is avoided.
- *Condensing pressure*. To avoid air infiltration in the cycle, a condensing pressure higher than atmospheric pressure is advisable. Water, however, has a condensing pressure generally lower than 100 mbar absolute. Low temperature organic fluids such as HFC-245fa, HCFC-123 or HFC-134a do meet this requirement. Organic fluids with a higher critical temperature on the other hand, such as hexane or toluene, are subatmospheric at ambient temperature.
- *Fluid characteristics*. Water as a working fluid is very convenient compared to organic fluids. Its main assets are low cost and high availability, non-toxicity, non-flammability, environmentally friendly (low Global Warming Potential and null Ozone Depleting Potential), chemical stability (no working fluid deterioration in case of hot spot in the evaporator), and low viscosity (and thus lower friction losses and higher heat exchange coefficients). However, steam cycles are in general not fully tight: water is lost as a result of leaks, drainage or boiler blow-down. Therefore, a water-treatment system must be integrated with the power plant to feed the cycle with high-purity deionized water. A deaerator must also be included to avoid corrosion of metallic parts due the presence of oxygen in the cycle.
- *Turbine design*. In steam cycles, the pressure ratio and the enthalpy drop over the turbine are both very high. As a consequence, turbines with several expansion stages are commonly used. In ORC cycles, the enthalpy drop is much lower, and single or two-stage turbines are usually employed, entailing lower cost.
- Additional consequences of the lower enthalpy drop of organic fluids include lower rotating speeds and lower tip speed. A lower rotating speed allows direct drive of the electric generator without reduction gear (this is especially advantageous for low power-range plants), while the low tip speed decreases the stress on the turbine blades and simplifies their design.
- *Efficiency*. The efficiency of current high temperature Organic Rankine Cycles does not exceed 24%. Typical steam Rankine cycles show a thermal efficiency higher than 30%, but with a more complex cycle design (in terms of number of components or size).

Table 3
Advantages and drawbacks of each technology.

Advantages of the ORC	Advantages of the steam cycle
No superheating	Higher efficiency
Lower turbine inlet temperature	Low-cost working fluid
Compactness (higher fluid density)	Environmental-friendly working fluid
Lower evaporating pressure	Non-flammable, non-toxic working fluid
Higher condensing pressure	Low pump consumption
No water-treatment system and deareator	High chemical-stability working fluid
Turbine design	
Low temperature heat recovery, once-through boiler	

The advantages of each technology are listed in Table 3.

In summary, the ORC cycle is more interesting in the low to medium power range (typically less than a few MWe), since small-scale power plants cannot afford an on-site operator, and because it requires simple and easy to manufacture components and design. It is consequently more adapted to decentralized power generation. For high power ranges, the steam cycle is generally preferred, except for low temperature heat sources [37].

5. Working fluid selection

The selection of working fluids has been treated in a large number of scientific publications. In most cases, these studies present a comparison between a set of candidate working fluids in terms of thermodynamic performance and based on a thermodynamic model of the cycle.

When selecting the most appropriate working fluid, the following guidelines and indicators should be taken into account:

(1) Thermodynamic performance: the efficiency and/or output power should be as high as possible for given heat source and heat sink temperatures. This performance depends on a number of interdependent thermodynamic properties of the working fluid: critical point, acentric factor, specific heat, density, etc. It is not straightforward to establish an optimum

for each specific thermodynamic property independently. The most common approach consists in simulating the cycle with a thermodynamic model while benchmarking different candidate working fluids.

(2) Positive or isentropic saturation vapor curve: as previously detailed in the case of water, a negative saturation vapor curve (“wet” fluid) leads to droplets in the later stages of the expansion. The vapor must therefore be superheated at the turbine inlet to avoid turbine damage. In the case of a positive saturation vapor curve (“dry” fluid), a recuperator can be used in order to increase cycle efficiency. This is illustrated in Fig. 9 for isopentane, R11 and R12.

(3) High vapor density: this parameter is of key importance, especially for fluids showing a very low condensing pressure (e.g. silicon oils). A low density leads to a higher volume flow rate: the sizes of the heat exchangers must be increased to limit the pressure drops. This has a non-negligible impact on the cost of the system. It should however be noted that larger volume flow rates can allow a simpler design in the case of turboexpanders, for which size is not a crucial parameter.

(4) Low viscosity: low viscosity in both the liquid and vapor phases results in high heat transfer coefficients and low friction losses in the heat exchangers.

(5) High conductivity is related to a high heat transfer coefficient in the heat exchangers.

(6) Acceptable evaporating pressure: as discussed for the case of water as working fluid, higher pressures usually lead to higher investment costs and increased complexity.

(7) Positive condensing gauge pressure: the low pressure should be higher than the atmospheric pressure in order to avoid air infiltration into the cycle.

(8) High temperature stability: unlike water, organic fluids usually suffer chemical deterioration and decomposition at high temperatures. The maximum heat source temperature is therefore limited by the chemical stability of the working fluid.

(9) The melting point should be lower than the lowest ambient temperature through the year to avoid freezing of the working fluid.

(10) High safety level: safety involves two main parameters—toxicity and flammability. The ASHRAE Standard 34 classifies refrigerants in safety groups and can be used for the evaluation of a particular working fluid.

(11) Low Ozone Depleting Potential (ODP): the ozone depleting potential is 11, expressed in terms of the ODP of the R11, set to unity. The ODP of current refrigerants is either null or very close to zero, since non-null ODP fluids are progressively being phased out under the Montreal Protocol.

(12) Low Greenhouse Warming Potential (GWP): GWP is measured with respect to the GWP of CO2, chosen as unity. Although some refrigerants can reach a GWP value as high as 1000, there is currently no direct legislation restricting the use of high GWP fluids.

(13) Good availability and low cost: fluids already used in refrigeration or in the chemical industry are easier to obtain and less expensive.

While fluid selection studies in the scientific literature cover a broad range of working fluids, only a few fluids are actually used in commercial ORC power plants. These fluids are summarized in Table 4, classified in terms of critical temperature [32].

In general, the selected fluid exhibits a critical temperature slightly higher than the target evaporation temperature: if the evaporation temperature is much higher than the critical temperature—for example if toluene ($Tc = 319$ °C) is evaporated at 100 °C—vapor densities become excessively low in both the high and low pressure lines.

Table 5 summarizes the scientific literature in the field of working fluid selection for ORC systems. To compare the different papers, three characteristics are taken into account: the target application and the considered condensing/evaporating temperature ranges. The papers comparing the working fluid performance as a function of the turbine inlet pressure (for example [57]) and not the temperature are excluded since the main limitation in the ORC technology is the heat source temperature and not the high pressure.

From Table 5 it becomes apparent that, despite the multiplicity of working fluid studies, no single fluid has been identified as optimal for the ORC. This is due to the different hypotheses used to perform fluid comparisons:

- Some authors consider the environmental impact (ODP, GWP), the flammability, and the toxicity of the working fluid, while other authors do not.

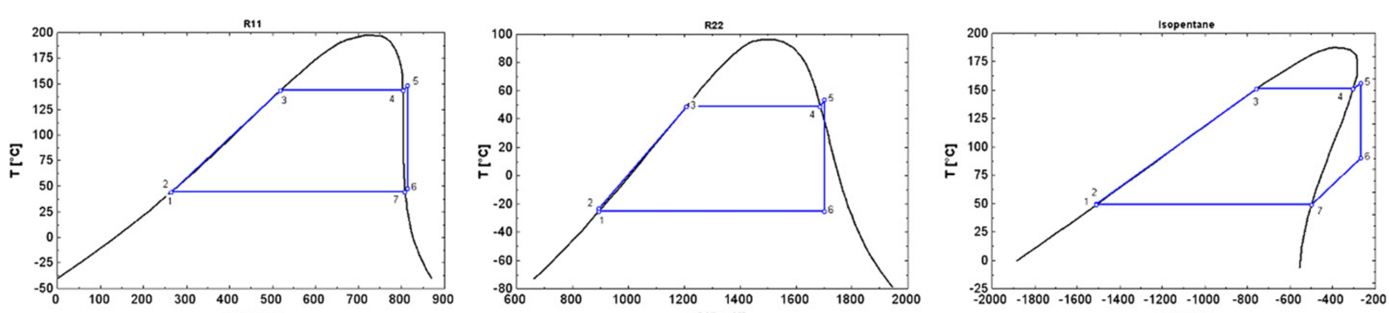


Fig. 9. Isentropic, wet and dry working fluids.

Table 4
Common working fluids in commercial ORC installations.

HFC-134a	Used in geothermal power plants or in very low temperature waste heat recovery.
HFC-245fa	Low temperature working fluid, mainly used in waste heat recovery.
n-pentane	Used in the only commercial solar ORC power plant in Nevada. Other applications include waste heat recovery and medium temperature geothermy.
Solkatherm	Waste heat recovery
OMTS	Biomass-CHP power plants
Toluene	Waste heat recovery

Table 5
Summary of different working fluid studies.

Ref.	Application	T_{cd} (°C)	T_{ev} (°C)	Considered fluids	Recommended fluids
[38]	WHR	30–50	120	R11, R113, R114	R113
[39]	n/a	35–60	80–110	Unconventional working fluids	HCFC-123, R124
[40]	WHR	30	150–200	HCFC-123, iso-pentane, HFE 7100, Benzene Toluene, p-xylene	Benzene, Toluene, HCFC-123
[17]	ICE	55 (100 for water)	60–150 (150–260 for water)	Water, HCFC-123, isopentane, R245ca, HFC-245fa, butane, isobutene and *R*-152a	Water, R245-ca and isopentane
[41]	CHP	90*	250–350*	ButylBenzene, Propyl-benzene, Ethylbenzene, Toluene, OMTS	ButylBenzene
[42]	Geoth.	30*	70–90	Ammonia, *n*-Pentane, HCFC-123, PF5050	Ammonia
[43]	WHR	35	60–100	HFC-245fa, HCFC-123, HFC-134a, *n*-pentane	HCFC-123, *n*-pentane
[44]	Geoth.	30	100	alkanes, fluorinated alkanes, ethers and fluorinated ethers	RE134, RE245, R600, HFC-245fa, R245ca, R601
[45]	Geoth	25	80–115	propylene, R227ea, RC318, R236fa, ibutane, HFC-245fa	Propylene, R227ea, HFC-245fa
[46]	WHR	25	100–210	R113, 123, R245ca, Isobutane	R113
[47]	Solar	35	60–100	Refrigerants	R152a, R600, R290
[48]	Solar	45	120/230	Water , *n*-pentane HFE 7100, Cyclohexane, Toluene , HFC-245fa , *n*-dodecane, Isobutane	*n*-dodecane
[49]	WHR	25	145*	water, ammonia, butane, isobutane R11 , HCFC-123, R141B, R236EA , R245CA , R113	R236EA
[50]	WHR	40	120	Alcanes, Benzene, R113 , HCFC-123 , R141b, R236ea, R245ca , HFC-245fa , R365mfc, Toluene	Toluene, Benzene
[51]	WHR	50	80–220	R600a, HFC-245fa, HCFC-123, R113	R113, HCFC-123
[52]	CHP	50	170	R365mfc, Heptane, Pentane, R12, R141b, Ethanol	Ethanol
[53]	ICE WHR	35	96–221	HFC-134a, R11, Benzene	Benzene
[54]	n/a	30	50–140	RC-318, *R*-227ea, *R*-113, iso-butane, *n*-butane, *n*-hexane, iso-pentane, neo-pentane, *R*-245fa, *R*-236ea, C5F12, R236fa	*n*-hexane
[55]	WHR	27–87	327*	HFC-245fa , R245ca ,R236ea, R141b , HCFC-123,R114, R113, R11, Butane	R11, R141b, R113, HCFC-123, HFC-245fa, R245ca
[56]	WHR	n/a	277*	R12, HCFC-123, HFC-134a, R717	HCFC-123
[13]	Solar	~30	150	*n*-Pentane, SES36, R245fa, R134a	R245fa, SES36

The part of the study evaluating supercritical working fluids has not been taken into account.

* Max/min temperature of the heat source/sink instead of evaporating or condensing temperature.

- Different working conditions (e.g. the considered temperature ranges) are assumed, leading to different optimal working fluids.
- The objective functions in the optimization depend on the target application: in CHP or solar applications the cycle efficiency is usually maximized, while in WHR applications, the output power should be maximized [58].

It follows that, since no working fluid can be flagged as optimal, the study of the working fluid candidates should be integrated into the design process of any ORC system.

Many studies [38,39,41,43,46,48,50–56] recommend the fluid with the highest critical temperature, which might suggest that the plant efficiency could be further improved by selecting even higher critical point working fluids [40]. However, as aforementioned, a high critical temperature also implies working at low vapor densities, leading to higher system cost.

It can therefore be concluded that the thermodynamic efficiency alone cannot be considered as the sole criterion for the selection of the working fluid. More holistic selection methods should be considered. However, very few studies include additional parameters taking into account the practical design of the ORC system, mainly because of the difficulty to define a proper function for a multi-objective optimization of the cycle. Examples of such studies are provided in [58–62], where a fluid selection taking into account the required heat exchange area, turbine size, cost of the system, risk, etc. are provided. These studies reveal that taking the economics into account can lead to the selection of very different optimal operating conditions and working fluids. Those methods should therefore be preferred to the simplistic thermodynamic benchmarking of candidate working fluids.

6. Expansion machines

The performance of an ORC system strongly correlates with that of the expander. The choice of the technology depends on the operating conditions and on the size of the system. Two main types of machines can be distinguished: the turbo and positive displacement types. Similar to refrigeration applications, displacement type machines are more appropriate in small-scale ORC units (Fig. 10), as they are characterized by lower flow rates, higher pressure ratios and much lower rotational speeds than turbo-machines [63].

6.1. Turbomachines

A distinction is generally made between two main types of turbines: the axial turbine and the radial inflow turbine.

Axial turbines show a distinct design when used in combination with high molecular weight working fluids. The main difference between organic fluids and steam is the enthalpy drop during the expansion, which is much higher for steam. As already mentioned, fewer stages are required in the case of an organic fluid. Even single-stage turbines can be employed for low or medium temperature ORC cycles.

Another characteristic of organic fluids is the low speed of sound. As a result, this speed is reached much sooner in an ORC than in a steam cycle and constitutes an important limitation as high Mach numbers are related to higher irreversibilities and lower turbine efficiencies.

Radial inflow turbines are designed for high pressure ratios and low working fluid flow rates. Their geometry allows higher peripheral speeds than for axial turbines, and therefore a higher enthalpy drop per stage. They also have the advantage of maintaining an acceptable efficiency over a large range of part-load

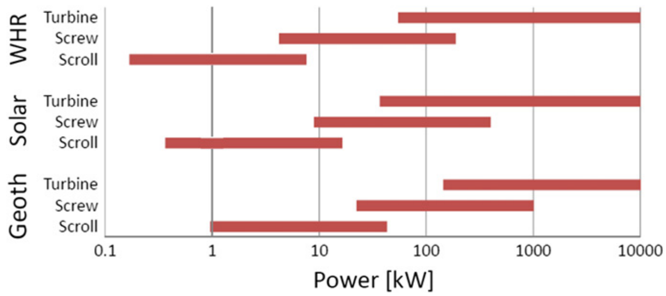


Fig. 10. Optimum operating map for 3 expander technologies and 3 target applications [62].

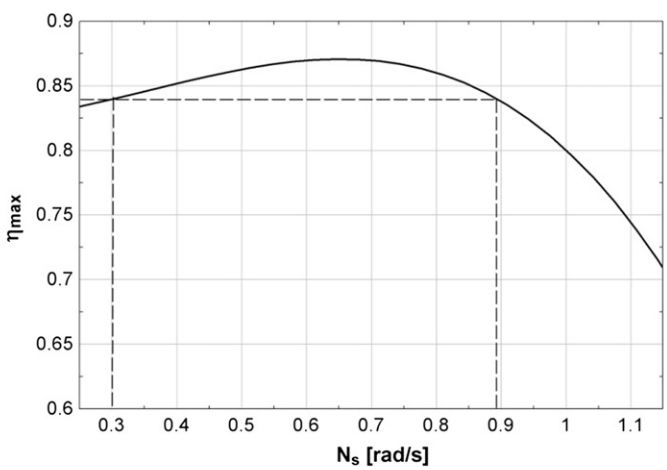


Fig. 11. Maximum radial turbine efficiency as a function of the specific speed.

conditions. However, unlike the axial turbine, it is difficult to assemble several stages in series. Fig. 11 shows a typical maximum efficiency curve as a function of the specific speed for a radial turbine. The specific speed is defined by

$$N_s = \frac{2\pi N\sqrt{\dot{V}_{ex}}}{\Delta h^{0.75}} \tag{1}$$

This maximum efficiency is the design point efficiency. It is obtained only when the speed triangles (i.e. the blade angles) are optimized for the design conditions. If an efficiency of 84% is required, the acceptable specific speed range is comprised between 0.3 and 0.9 for the particular case of Fig. 11.

Turbomachines are not suitable for very small-scale units, mainly because their rotating speed increases dramatically with decreasing turbine output power. This is due to a typical characteristic of turbomachines: for a given technology, their tip speed is approximately constant, independent of the turbine size. This tip speed can be written

$$U_2 = \pi N D_2 \tag{2}$$

where U_2 is the tip speed, N is the rotating speed and D_2 is the outer diameter.

As a consequence, when the turbine size (D_2) decreases, the rotating speed increases proportionally [63]. This high rotating speed is the main reason why micro-scale turbomachines are not yet available on the market. It should however be noted that some lab-scale prototypes have been successfully developed and tested:

- Pei et al. [64] tested a 600 W radial turbo-expander with gearbox using air and reached 42% efficiency at 55,000 rpm;
- Kang [65] developed a 30 kW radial turbine for HFC-245fa using a high-speed generator (20,000 rpm). Maximum electrical overall isentropic efficiency was about 67% (value recalculated from plots).

6.2. Positive displacement expanders

The major types of positive displacement expanders are piston, scroll, screw and vane expanders. In piston expanders, the same volume functions successively as the suction, expansion and discharge chamber according to the timing of the suction and discharge valves. In rotary expanders (scroll, screw, vanes), those chambers co-exist. The suction chamber evolves into one or two expansion chambers (for instance scroll expanders are characterized by two expansion chambers) after one shaft revolution. Similarly, expansion chambers become discharge chambers once they get into contact with the discharge line of the machine.

In contrast with most piston expanders, rotary expanders do not need valves: the timing of the suction and discharge processes is imposed by the geometry of the machines. In terms of design, this is a major advantage over piston expanders. Moreover, the fact that suction and discharge do not occur in the same location limits the suction heat transfer, which has a positive impact on the volumetric performance of the machine. On the other hand, piston expanders typically show lower internal leakage than scroll and screw expanders.

While technically mature turbomachines are available on the market for large ORC units, almost all positive displacement expanders that have been used up to now are prototypes, often derived from existing compressors [66–69]. Positive displacement expanders are a good substitute for turbomachines at low output powers: their rotating speed is limited (generally 1500 or 3000 rpm on a 50 Hz electrical grid), they are reliable (widely used for compressor applications), they tolerate the presence of a liquid phase during expansion, and they exhibit good isentropic efficiency.

In such a machine, the decrease of the pressure is caused by an increase of the volume of the expansion chambers. The ratio between the volume of the expansion chamber(s) at the end of the expansion and that at the beginning is called "built-in volume ratio" ($r_{v,in}$). The expansion process is illustrated in Fig. 12 for the particular case of a scroll expander: fluid is admitted at the center and trapped in a pocket that is progressively expanded while traveling to the periphery, where the working fluid is finally discharged.

Two types of losses can occur if the system specific volume ratio is not equal to the expander nominal volume ratio (Fig. 13):

- Under-expansion occurs when the internal volume ratio of the expander is lower than the system specific volume ratio. In that case, the pressure in the expansion chambers at the end of the expansion process (P_{in}) is higher than the pressure in the discharge line.
- Likewise over-expansion occurs when the internal volume ratio imposed by the expander is higher than the system specific volume ratio.

These two effects can considerably reduce the efficiency of the expansion process, the most common being under-expansion. As a consequence, volumetric expanders are generally less adapted to high expansion ratios than turbomachines. Other sources of losses include friction, supply pressure drop, internal leakage and heat transfers [69].

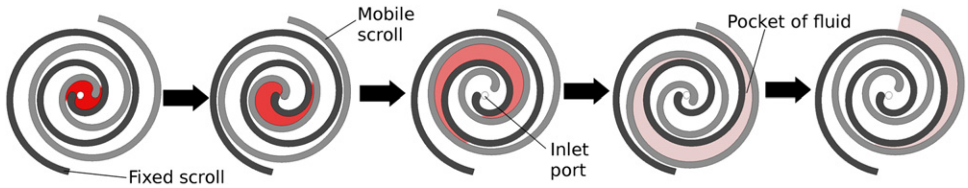


Fig. 12. Operating principle of a scroll expander.

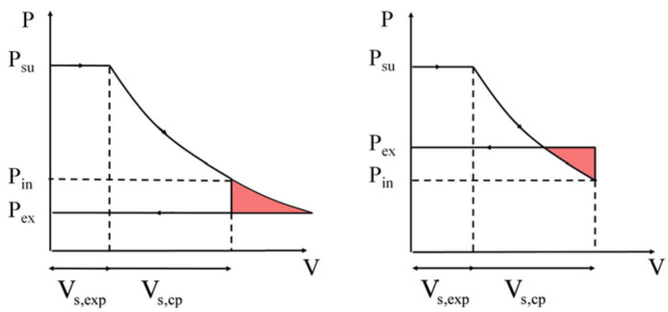


Fig. 13. Under (left) and over (right) expansion losses.

To optimize the performance of the expander and to minimize under-expansion and over-expansion losses, this built-in volume ratio should match the operating conditions. However, volume expansion ratios achieved in Rankine cycle systems are typically larger than those achieved in vapor compression refrigeration systems, which justifies the development of adapted designs of such expanders, rather than retrofitting existing compressors. Generally speaking, piston expanders are more appropriate for applications with large expansion ratios because their design allows for higher internal built-in volume ratios.

A major difficulty associated with the use of a positive displacement machine is its lubrication. One solution consists in installing an oil separator at the expander exhaust. In this case, unlike with compressors, an oil pump is necessary to drive the separated oil back to the expander suction. Another solution consists in circulating the oil with the refrigerant through the cycle and to install an oil separator at the evaporator exhaust. Separated oil is injected into the bearings, while the lubrication of the two spirals (in the case of a scroll expander) relies on the slight inefficiency of the separator. Alternatively, oil-free machines can be used, but these generally exhibit lower volumetric performance and high leakage due to larger tolerances between moving parts [67,69].

In some operating conditions (wet fluids with limited superheating at the expander supply), liquid may appear at the end of the expansion. This could pose a damage threat for piston expanders, but not for scroll and screw expanders, since the latter can generally accept a large liquid mass fraction.

6.2.1. Performance indicators for positive-displacement expanders

The literature review regarding the performance of volumetric expander prototypes reveals that different performance indicators are in use. Some authors [68,70–73] define the isentropic efficiency as the ratio between the measured enthalpy difference and the isentropic enthalpy difference, while some others [66,69,74] define it as the ratio of the measured output power divided by the isentropic expansion power:

$$\varepsilon_{s,1} = \frac{h_{su}-h_{ex}}{h_{su}-h_{ex,s}}; \quad \varepsilon_{s,2} = \frac{\dot{W}}{\dot{M}(h_{su}-h_{ex,s})} \tag{3}$$

The difference between the two definitions depends on the ambient heat losses and can be obtained by performing an energy balance over the expander:

$$\varepsilon_{s,2} = \frac{\dot{W}}{\dot{M}(h_{su}-h_{ex,s})} = \frac{\dot{M}(h_{su}-h_{ex})-\dot{Q}_{amb}}{\dot{M}(h_{su}-h_{ex,s})} = \varepsilon_{s,1} - \frac{\dot{Q}_{amb}}{\dot{M}(h_{su}-h_{ex,s})} \tag{4}$$

where $\dot{W}$ is the output power, h_{su} is the supply enthalpy, h_{ex} is the exhaust enthalpy) and $\dot{M}$ (the mass flow rate) are measured values. $\dot{Q}_{amb}$ is the ambient heat loss and $h_{ex,s}$ is the isentropic exhaust enthalpy.

The isentropic efficiency defined as the enthalpy ratio ($\varepsilon_{s,1}$) should be used for adiabatic processes only (i.e. ambient heat losses are neglected). However, according to [74], volumetric expanders, even insulated, release a non-negligible amount of heat to their environment. This can lead to biased values of the measured isentropic efficiency: if an expander produces 0.7 kW of shaft power for an isentropic expansion power ($\dot{M}.(h_{su}-h_{ex,s})$) of 1 kWe, the efficiency is ε_s=70%. However, if this expander also exchanges 0.5 kW of thermal power with its environment, the “enthalpy ratio” definition of the isentropic efficiency leads to $\varepsilon_{s,1}$=120%, which is obviously erroneous.

Therefore, in order to provide a fair comparison between different reported efficiencies, the more general definition should be used. This is the one selected for the further developments of this paper:

$$\varepsilon_s = \frac{\dot{W}}{\dot{M}(h_{su}-h_{ex,s})} \tag{5}$$

Another difference between reported efficiencies lies in the type of output power: electrical or mechanical. Mechanical isentropic efficiencies are usually used for open-drive expanders while electrical isentropic efficiencies are used for hermetic expanders (in which the shaft is not accessible). The difference between both definitions is the generator efficiency (usually between 80% and 95%). Therefore, the type of reported efficiency should always be provided.

A second indicator must also be defined to account for the volumetric performance of the machine. In compressor mode, such indicator is called volumetric efficiency. In expander mode, this number can be higher than one because of internal leakage [74]. Therefore, a different variable name, the filling factor, is used. It is defined by

$$\varphi = \frac{\dot{M} v_{su}}{\dot{V}_s} \tag{6}$$

6.2.2. Reported performance

Table 6 summarizes the reported performance in experimental studies on volumetric expanders. The selected performance indicators are the mechanical/electrical isentropic efficiency (Eq. (5)) and the filling factor. Note that because of divergences in the definitions of these indicators in the papers, some efficiencies have been recalculated or evaluated from plots. In these studies, the best performance was achieved with scroll expanders, with mechanical efficiencies higher than 70% and electrical efficiencies

Table 6
Overview of previous experimental studies on positive-displacement expanders.

Ref.	Expander specs	Testing conditions	Max achieved performance
Badr et al. [38]	Rotary vane expander	$fluid = R113$ $p_{su} < 7$ bar	$\varepsilon_{s,mech} = 55\%$ $\dot{W}_{mech} = 1.6$kW
Zanelli and Favrat [66]	Hermectic, lubricated scroll expander	$T_{su} = 170°C$ $p_{su} = 6.4$bar; $N_{rot} = 2400–3600$rpm; $fluid = R134a$ $r_p = 2.4–4$	$\varepsilon_{s,el} = 65\%$ $\varphi = 95–120\%$ $\dot{W}_{el} = 1–3.5$kW
Yanagisawa et al. [67]	Oil-free scroll air machine $V_{s,cp} \approx 100$cm^3 $r_{v,nin} = 3.18$	$p_{su} = 6.5$bar $N_{rot} = 2500$rpm; $fluid = air$	$\varepsilon_{s,mech} = 60\%$ $\varphi = 76\%$
Manzagol et al. [76]	Cryogenic scroll expander, $\dot{V}_{s,} = 10$l/h	$p_{su} = 7$bar $T_{su} = 35$K $fluid = Helium$	$\varepsilon_{s,mech}=60\%$
Kane et al. [77]	Hermectic, lubricated scroll expander $r_{v,nin}=2.3$	$N_{rot} = 3000$rpm	$\varepsilon_{s,el} = 68\%$ $\dot{W}_{el} = 6.5$kW
Ingley et al. [78]	Scroll expander	$N_{rot}=2000$ rpm; $fluid=amomnia$	$\varepsilon_{s,mech}=18.2\%$ $\dot{W}_{mech} = 0.209$kW
Xiaojun et al. [79]	Scroll expander (fuel cell)	$p_{su} = 1–4$bar; $fluid = air$	$\varepsilon_{s,mech} = 69\%$ $\dot{W}_{mech} = 3.5$kW
Aoun and Clodic [68]	Oil-free scroll air machine $V_{s,ext} = 31.5$cm^3 $r_{v,nin} = 3.18$	$T_{su} = 190°C$ $N_{rot} = 1600–2500$rpm; $fluid = steam$ $r_p = 3–5$	$\varepsilon_{s,2,mech}=48\%$[a] $\varphi = 62\%$ $\dot{W}_{mech} = 500$W
Peterson et al. [75]	Kinematically rigid scroll expander $V_{s,exp} = 12$cm^3 $r_{v,nin} = 4.57$	$T_{su} = 170°C$ $p_{su} = 6.4$ bar; $N_{rot} = 1287$rpm; $fluid = R123$ $r_p = 3.82$	$\varepsilon_{s,mech} = 49.9\%$ $\varphi = 40–50\%$ $\dot{W}_{mech} = 0.256$kW
Kim et al. [80]	Self-designed double-sided scroll expander	$fluid = steam$ $p_{su} = 13.8$bar $N_{rot} = 1000–1400$rpm	$\varepsilon_{s,mech} = 34\%$ $\varphi = 0.42–0.52$ $\dot{W}_{mech} = 11.5$kW
Saitoh et al. [81]	Scroll expander	$fluid = R113$ $p_{su} = 9.4$bar; $T_{su} = 136°C$; $N_{rot} = 1800$rpm	$\varepsilon_{s,mech}=59.6\%$[b] $\dot{W}_{mech} = 450$W
Mathias et al. [82]	Refrigeration Scroll expander	$fluid = R123$ $r_p = 3–8.3$	$\varepsilon_{s,el}=48.3\%$[b] $\dot{W}_{el} = 2.9$kW
Mathias et al. [82]	Gerotor expander	$fluid = R123$ $r_p = 3–8.3$	$\varepsilon_{s,mech} = 35.1\%$ $\dot{W}_{mech} = 1.8$kW
Lemort et al. [69]	Oil-free open-drive scroll air machine $r_{v,nin} = 4$ $V_{s,exp} = 36.54$ cm^3	$T_{su} = 143°C$ $p_{su} = 10$bar; $fluid = R123$	$\varepsilon_{s,mech} = 68\%$ $\varphi = 1–1.34$ $\dot{W}_{mech} = 1.8$kW
Wang et al. [83]	Compliant scroll expander derived from an existing compressor $V_{s,exp} = 6.5$cm^3 $r_{v,nin} = 2.5$	$T_{su} = 125°C$; $p_{su} = 10–18$ bar; $N_{rot} = 2005–3670$rpm; $fluid = R134a$ $r_p = 2.65–4.84$	$\varepsilon_{s,mech} = 77\%$ $\dot{W}_{mech} = 1$kW
Manolakos et al. [70]	Automotive A/C scroll expander	$fluid = R134a$ $N_{rot} = 891$rpm	$\varepsilon_{s,mech} < 50\%$
Harada [84]	Refrigeration scroll expander with direct shaft connection	$fluid = R245fa$ $r_p = 2–7$;	$\varepsilon_{s,mech} = 87\%$ $\dot{W}_{mech} = 1$kW

Table 6 (*continued*)

Ref.	Expander specs	Testing conditions	Max achieved performance
Wei et al. [85]	Single screw expander	$T_{su}=17^{\circ}C$ $p_{su}=6$ bar; $p_{su}=6$ bar; $N_{rot}=2850$rpm; $fluid=air$	$\varepsilon_{s,mech}=30.76\%$ $\dot{W}_{mech}=5$kW
Lemort et al. [74]	Hermetic refrigeration scroll machine $r_{v,in}\approx 3$	$fluid=R245fa$ $r_p=2-6$ $T_{su}=92-140\ ^{\circ}C$; $p_{su}=6-16$bar;	$\varepsilon_{s,el}=68\%$ $\varphi=1-1.1$ $\dot{W}_{el}=2.2$kW
Schuster [86]	Screw expander	Fluid = R245fa $p_{su}=8-9$ bar; $N_{rot}=500-2300$ rpm	$\varepsilon_{s,el}=60\%$ $\dot{W}_{el}=2.5$kW
Qiu et al. [87]	Compressed-air-driven vane-type air expander	$fluid=HFE7000$ $p_{su}=6.7$ bar $T_{su}=117\ ^{\circ}C$; $N_{rot}=1689$ rpm.	$\varepsilon_{s,el}$=26%[b] $\dot{W}_{el}=850$W
Melotte [88]	Single-screw expander	$fluid=Solkatherm$ $p_{su}=5-10.5$ bar; $r_p=6.3-10.2$ $N_{rot}=3000$ rpm	$\varepsilon_{s,el}$=60.1[b] $\dot{W}_{el}=8.6$kW

[a] The authors only provided the "enthalpy ratio" definition of the isentropic efficiency (Eq. (4)).
[b] Recalculated with the measured values provided in the paper.

higher than 60%. Screw expander performance seems to be slightly lower: none of the experimental studies reports performance much higher than 60%. Other types of positive-displacement expanders such as the vane or gerotor types also show low efficiencies, which may be due to the fact that experimental studies on these machines are scarce in the literature.

7. Heat exchangers

The remaining components of the cycle, although well-known, also deserve some attention, particularly in light of their selection and integration into ORC systems. Heat exchangers represent a major share of the total module cost. Their optimization should therefore be carefully performed.

Key characteristics regarding heat transfers are the efficiency (or pinch point) and pressure drops. Each heat exchanger in the cycle is sized according to these two parameters. Different types of heat exchangers can be used, the most common being shell & tube (mainly in larger-scale systems) and plate heat exchangers (mainly in small-scale systems, due to their compactness).

A critical heat exchanger is usually the exchanger installed on the heat source. Depending on its nature, this heat exchanger must withstand high temperatures and can be subject to fouling and/or corrosion. In case of waste heat recovery, the heat exchanger must not interfere with the process, i.e. the pressure drop should be limited and its dimensions must comply with the available space. Moreover, in case of flue gases with sulfur content, the acid dew point should be avoided. This explains why in most commercial plants the exhaust gases are not cooled below 120–180 °C, depending on the sulfur content of the gases. Research is being carried out to design heat exchangers able to withstand and evacuate acid condensates.

Heat can be recovered by means of two different setups: (1) direct heat exchange between heat source and working fluid and (2) an intermediate heat transfer fluid loop that is integrated to transfer heat from the waste heat site to the evaporator, usually using thermal oil. Direct evaporation, although more efficient and conceptually simpler, involves a number of issues:

- At high temperatures (e.g. during start-up and transients), the working fluid can deteriorate when its maximal chemical stability temperature is reached, or when hot spots appear in the heat exchanger.
- The controllability and the stability of the systems are harder to achieve in case of direct evaporation. In contrast, a heat transfer loop damps the fast variations of the heat source and allows smoother cycle operation (e.g. to control the superheating in a solar ORC system).

As a consequence, most commercial ORC installations make use of an intermediate heat transfer loop (see for example Fig. 3).

Advanced architectures also focus on the integration of the heat exchangers. A good example is provided by the Turboden unit (Fig. 14): the condenser and the recuperator are integrated into a single component together with the turbine and the liquid receiver, which increases compactness, avoids piping and reduces leakages.

8. Pump

ORC feed pumps are key components and should be given particular care during the selection and sizing process. They should comply with the cycle requirement in terms of controllability, efficiency, tightness and NPSH, as described below.

8.1. Controllability

In most ORC systems, the pump is used to control the working fluid mass flow rate. The electrical motor is connected to an inverter which allows varying the rotating speed. In positive-displacement pumps, the flow rate is roughly proportional to the rotating speed, while in centrifugal pumps this flow rate also

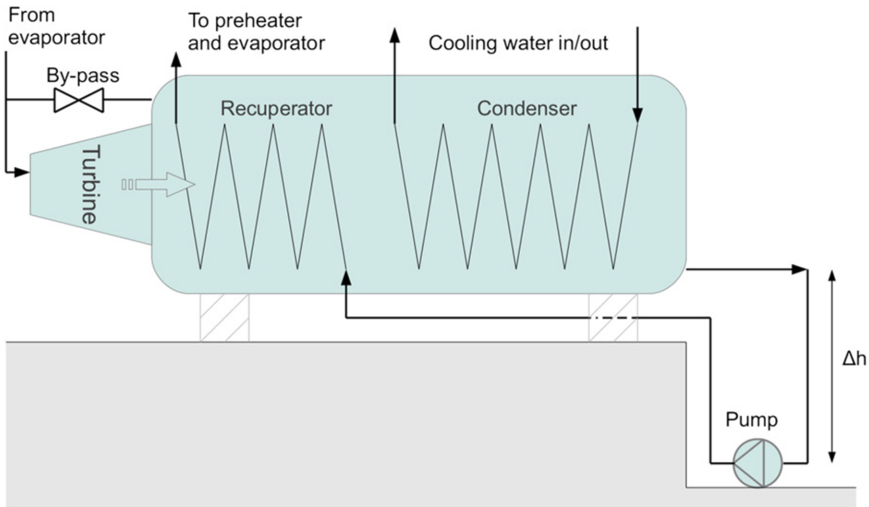


Fig. 14. Schematic view of the turboden turbine–recuperator–condenser assembly.

depends on the pressure head (i.e. the difference between evaporating and condensing pressure).

8.2. *Efficiency*

In traditional steam Rankine cycles, the pump consumption is very low compared to the output power. However, in an ORC cycle, the pump irreversibilities can substantially decrease the cycle overall efficiency. The ratio between pump electrical consumption and expander output power is called Back Work Ratio (BWR):

$$BWR = \frac{\dot{W}_{pp}}{\dot{W}_{exp}} \tag{7}$$

Fig. 15 shows the BWR for a few typical fluids as a function of the evaporating temperature. Two conclusions can be drawn from this figure:

(1) The higher the critical temperature of the working fluid, the lower the BWR.
(2) BWR increases with T_{ev}, and gets significantly high when operating the cycle close to the critical point.

Therefore, the pump efficiency is a crucial parameter in low temperature cycles and in transcritical cycles. Few pump efficiencies are reported in the literature, and they are usually quite low for low capacity units:

- An overall isentropic efficiency of 25% has been reported by some of the authors in a 2 kWe ORC unit [89].
- Reid [90] reports a pump efficiency of 7% on kW-scale ORC cycle using HFE-7000;
- Quoilin [91] obtained a 22% efficiency on a diaphragm pump using RHFC-245fa.
- Bala et al. [92] studied the influence of different working fluids on the overall efficiency of sliding-vane refrigerant pumps; the highest reported efficiency was about 20%.
- Melotte [88] performed an experimental study on a centrifugal pump working with solkatherm, and obtained an efficiency varying between 10% and 20%.

Note that these efficiencies are all electrical efficiencies, i.e. incorporate the motor efficiency, which can be low for small units or if the motor is oversized.

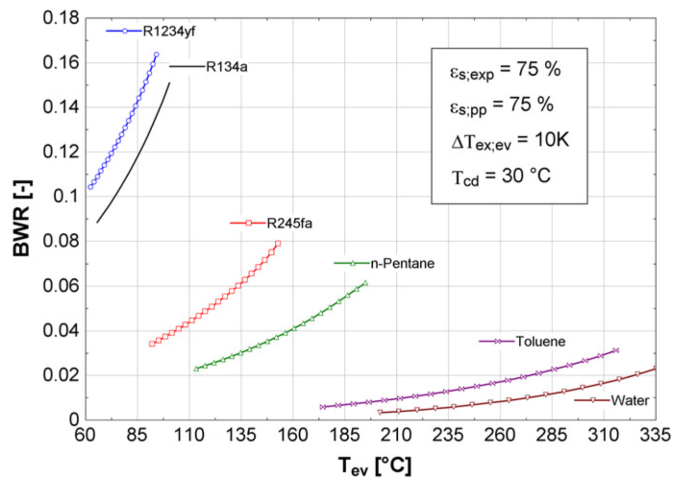


Fig. 15. BWR as a function of evaporation temperature for different fluids.

According to manufacturer data, centrifugal pumps (usually used in larger-scale units) should exhibit efficiencies higher than 60%, and diaphragm pump should operate over 40–50%. However, no actual data is available in the literature to confirm these figures.

8.3. *Tightness*

Organic fluids are expensive and can be flammable, toxic and have high GWP or ODP values. Hence, it is important to ensure full tightness of the cycle. This explains why diaphragm pumps are usually preferred to piston pumps. Note that diaphragm pumps generate a pulsed flow rate, which can, in some cases, constitute a drawback (e.g. because of fluctuations in pressure and flow rate measurements). When using centrifugal pumps, tightness is ensured by a shaft seal.

8.4. *Low net pressure suction head (NPSH)*

This parameter is critical for the design of the ORC. Strategies must be set up to avoid pump cavitation, which can lead to damages in the pump, to a reduction of the working fluid flow rate, and to the necessity to shut down the cycle. The most common strategies are briefly described in the next sections.

8.4.1. Pre-feed pump

A first pump with low NPSH is added before the main feed pump (see Fig. 16) to provide the required pressure head. This is the strategy selected e.g. in the Tri-O-Gen Unit, in which the main pump is directly connected to the turbine shaft and enclosed in a hermetic container with the generator to avoid leakage of toluene (Fig. 16). To avoid cavitation, the pressure head provided by the pre-feed pump must be higher than the main pump NPSH for all flow rates:

$$\Delta p_{pre-feed}(\dot{M}) > NPSH_{pp} \quad (8)$$

8.4.2. Gravity fed working fluid pump

The required pressure head can be provided by a static pressure difference due to the vertical distance between the condenser (or the liquid receiver) and the pump. This strategy was selected by companies such as Turboden (typically for large-scale ORC plants), which install the feed pumps in a cavity about 4 m lower than the condenser (Fig. 14). The no-cavitation condition can be written

$$\rho g \Delta h > NPSH_{pp} \quad (9)$$

where ρ is the fluid density in liquid state, g is the gravity and Δh is the level difference.

8.4.3. Addition of non-condensing gases

When adding non-condensable gases (e.g. nitrogen), a small fraction of them is dissolved in the working fluid and circulates through the cycle. However, the main part remains in gaseous state and is carried by the main flow to the condenser, where it accumulates because of the condensation of the working fluid. It is therefore in the condenser and in the liquid receiver that the concentration of non-condensable gases is highest.

The pressure at the pump inlet is the sum of two partial pressures:

→The partial pressure of non-condensing gases in the condenser and in the liquid receiver ($p_{part,ncg}$).
→The partial pressure of the working fluid in vapor phase, corresponding to the saturation pressure at the given temperature ($p_{part,wf}$).

$$p_{tot} = p_{part,ncg} + p_{part,wf} \quad (10)$$

When the working fluid leaves the gas–liquid interface toward the pump (i.e. it is no longer in contact with the non-condensing gases), its pressure is higher than the saturation pressure. It is therefore subcooled [93]. This subcooling is expressed in terms of the difference between the saturation temperature corresponding to the total pressure and the actual condensing temperature (i.e. the temperature measured in the condenser and at the pump inlet):

$$\Delta T_{sc} = T_{sat}(p_{tot}) - T_{cd} \quad (11)$$

The condition for no cavitation states that the amount of subcooling must be sufficient to provide the NPSH required by the pump, i.e. the partial pressure of the introduced non-condensing gases must be higher than NPSH:

$$\Delta p(\Delta T_{sc}) = p_{tot} - p_{part,wf} = p_{part,ncg} > NPSH_{pp} \quad (12)$$

8.4.4. Thermal subcooling

The required NPSH can be obtained by thermally subcooling the working fluid. As shown in Fig. 17, this can be performed in three different ways:

(A) Using an additional heat exchanger (subcooler) after the liquid receiver. It should be noted that a separate liquid receiver is not compulsory. The shell of the condenser (e.g. in case of shell and tube heat exchanger) can also play this role and absorb the level fluctuations of the working fluid.
(B) Adding the subcooler directly into the liquid receiver. This solution avoids the use of an additional heat exchanger. It is the one selected in the Eneftec unit.
(C) Subcooling in the condenser by ensuring that a part of it is flooded by liquid. This solution corresponds to a very simple architecture but is also the most difficult to control: the refrigerant charge must be exactly adapted to the required subcooling (too much refrigerant in the cycle entails a larger liquid zone in the condenser and therefore excessive subcooling) and the operating conditions cannot vary since this causes fluctuations of the liquid level.

The working fluid temperature at the outlet of the subcooling (Tsu,pp) system must comply with the following no-cavitation condition:

$$p_{cd} - p_{sat}(T_{su,pp}) > NPSH_{pp} \quad (13)$$

9. Next generation organic Rankine cycles and current R&D

At the present time, most commercial ORC plants exhibit a simple architecture: sub-critical working conditions, pure working fluids, single evaporation pressure, and possible use of a recuperator.

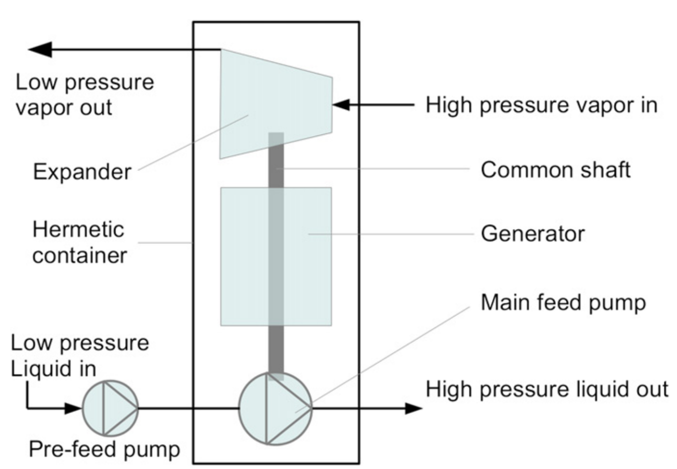


Fig. 16. ORC system with common pump and expander shaft and pre-feed pump.

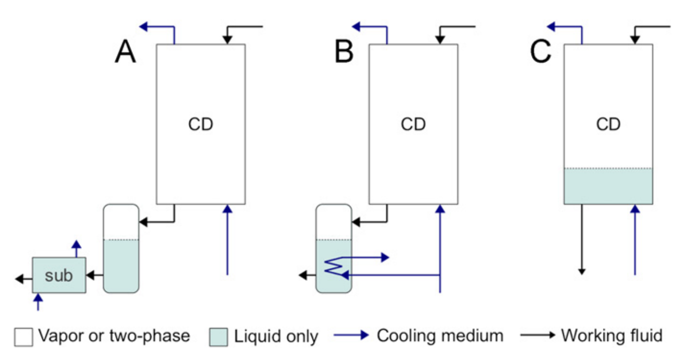


Fig. 17. Different strategies for thermal subcooling.

Table 7
Advanced architectures for the next generation Organic Rankine Cycles.

Architecture	Performance improvement (%)	Ref.
Transcritical cycles	8	[94]
Zeotropic mixtures	up to 16	[95]
Regenerative cycles	14	[96]
Cascaded Cycles	N/A	[77]
Cycles with reheating	4	[97]
Two-phase expansion cycles	N/A	[98]
Multiple evaporation pressures	16	[99]

While the current state of the art shows a maturity for the first generation of ORC cycles, there is still an important lack of know–how and room for improvement, urging for further strategic basic research. Analogous to the historical improvement of the steam cycle efficiency (e.g. about 20% with the introduction of the supercritical cycle), the goal should be to increase the ORC efficiency (typically 16%) beyond 20%.

Current R&D focuses on working fluid selection issues (see Table 5 for a review), but also on innovative cycles architectures. The main investigated tracks are summarized in Table 7, with their respective potentials for performance increase.

Some research groups focus on turbine optimization, which involves studying real-gas effects (in particular close to the critical point) and developing new accurate equations of states (see e.g. [100]).

Regarding the control strategies, state-of-the-art ORC units are usually designed for a nominal operating point, and exhibit poor performance in part-load conditions. In a previous work [101], it was demonstrated that the cycle second-law efficiency can be improved by about 10% by implementing a proper control strategy taking into account the heat source variability, and by continuously re-optimizing the operating conditions.

10. Conclusions

In this paper, the current state of the ORC technology was described, with an emphasis on the temperature levels and on the specificities of each application. The main manufacturers were listed, describing their activity field, the main technological characteristics of their ORC solutions, and their power range.

Comparison with the traditional steam cycle revealed that ORC cycles are more appropriate for moderate power ranges and/or for low-temperature application such as low temperature waste heat recovery or geothermal.

The ORC market has grown exponentially since the beginning of the 1980s, mainly in the fields of biomass CHP, geothermal energy and waste heat recovery. A compilation of the available market data showed that actual plants size is limited principally by a minimum power output of a few hundreds of kWe. Low-capacity systems are currently under development or in the demonstration phase but still require niche markets to begin industrial production and reduce their cost.

Working fluids and expansion machines are two key aspects of ORC technology. This survey underlined the large number of working fluid studies in the literature and pointed out their limitations. The need for more holistic working fluid studies, taking into account additional properties of the working fluid beyond the sole thermodynamic performance was highlighted. Studies taking into account the impact of the working fluid on the system cost or on the component size are of particular interest and constitute a research area that should be further explored.

Positive displacement machines are preferably used for small-scale applications. At present, most of the employed positive displacement expanders are obtained by modifying existing compressors. Turbomachines are mainly designed for larger-scale applications and show a higher degree of technical maturity.

The literature review of small-scale expanders showed that the scroll expander is the most widely used for very small-scale applications, while screw expanders are used for slightly higher output powers (up to a few hundreds kWe). Reported overall isentropic efficiencies are very variable, with a maximum value around 70% if generator losses are included, or 80% for the mechanical isentropic efficiency. The need for a unified definition of the performance indicators in experimental studies was highlighted, and a general definition of the isentropic efficiency was proposed.

In the last part of the paper, more particular technological issues such as pump cavitation or acid dew point were considered, describing the most common solutions proposed by manufacturers or in the scientific literature.

Finally, the current tracks for R&D were described, providing figures of their potential impact on the cycle performance.

Acknowledgments

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